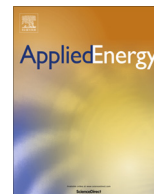




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Development of a 3 kW double-acting thermoacoustic Stirling electric generator

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HIGHLIGHTS

- A 3 kW double-acting thermoacoustic Stirling electric generator is introduced.
- 1.57 kW electric power with 16.8% thermal-to-electric efficiency was achieved.
- High mechanical damping coefficient greatly decreases the system performance.
- Performance difference is significant, which also decreased system performance.

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ABSTRACT

In this paper, a double-acting thermoacoustic Stirling electric generator is proposed as a new device capable of converting external heat into electric power. In the system, at least three thermoacoustic Stirling heat engines and three linear alternators are used to build a multiple-cylinder electricity generator. In comparison with the conventional thermoacoustic electricity generation system, the double-acting thermoacoustic Stirling electric generator has advantages on efficiency, power density and power capacity. In order to verify the idea, a prototype of 3 kW three-cylinder double-acting thermoacoustic Stirling electric generator is designed, built and tested. Based on the classic thermoacoustic theory, numerical simulation is performed to obtain the thermodynamic parameters of the engine. And distributions of key parameters are presented for a better understanding of the energy conversion process in the engine. In the experiments, a maximum electric power of about 1.57 kW and a maximum thermal-to-electric conversion efficiency of 16.8% were achieved with 5 MPa pressurized helium and 86 Hz working frequency. However, we find that the mechanical damping coefficient of the piston is dramatically increased due to the deformation of the cylinder wall caused by high thermal stress during the experiments. Thereby, the system performance was greatly reduced. Additionally, the performance differences between three engines and three alternators are significant, such as the heating temperature difference between three heater blocks of the engines, the piston displacement and the output electric power differences between three alternators. These problems need further investigation. This work presents a new thermal-to-electric conversion technology, which can be utilized in many energy area, such as solar energy, industrial waste heat and so on.

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1. Introduction

Due to the high flexibility of heat source, the outer combustion engines attract significant attentions nowadays in solar energy application, waste heat utilization and other areas. Other than the conventional Stirling heat engine, the thermoacoustic Stirling

heat engines (TASHE) is another special type of such engine capable of converting external heat to acoustic power (i.e. mechanical work). The acoustic power can be further applied to produce electricity by driving linear alternator or realize the heat pumping from low temperature to high temperature by driving thermoacoustic refrigerator [1] and cryocooler [2]. In comparison with the conventional Stirling heat engine, the TASHE can not only achieve a high efficiency as the Stirling engine, but also provide much higher reliability by eliminating moving part within hot environment. The linear alternator based on the linear motor technology is capable

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Nomenclature

Abbreviation

DATASEG	double-acting thermoacoustic Stirling electric generator
TASEG	thermoacoustic Stirling electric generator
TASHE	thermoacoustic Stirling heat engine

Symbols

A	cross sectional area of flow channel (m^2)
A_s	cross sectional area of solid wall (m^2)
A_p	cross sectional area of piston (m^2)
c_p	isobaric heat capacity per unit mass (J/kg K)
d	diameter (mm)
f_k	spatially averaged thermal function
f_v	spatially averaged viscous function
H	total energy (W)
I	current amplitude (A)
$\text{Im}[\]$	imaginary part of complex number
i	$\sqrt{-1}$
K	spring stiffness (N/m)
k	thermal conductivity of gas (W/m K)
k_s	thermal conductivity of solid (W/m K)
L	electrical inductance of winding (H)
l	length (mm)
M	moving mass (kg)
PR	pressure ratio
P_0	mean pressure (Pa)
p	pressure amplitude (Pa)
p_b	pressure amplitude at the back volume of the alternator (Pa)
Q	heat exchange between gas and solid wall (W)

Q_h	heating power (W)
R	load resistance (Ω)
$\text{Re}[\]$	real part of complex number
R_m	mechanical damping coefficient (N s/m)
r	electrical resistance of winding (Ω)
T	temperature (K)
TM	transfer matrix
U	volume velocity amplitude (m^3/s)
W_a	acoustic power (W)
W_e	electric power (W)
x	displacement (m)
γ	adiabatic index
η_{ae}	acoustic-to-electric conversion efficiency
η_{te}	thermal-to-electric conversion efficiency
θ	phase angle ($^\circ$)
θ_{pU}	phase angle between pressure and volume velocity wave ($^\circ$)
θ_{VI}	phase angle between voltage and current ($^\circ$)
ρ	density (kg/m^3)
σ	Prandtl number
τ	transduction coefficient (N/A)
ω	angular frequency (rad/s)
$\ $	magnitude of complex number

Superscript

*	conjugation of complex number
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Subscript

c	compression
e	expansion

of converting the acoustic power to electric power. By employing flexure bearing suspension technology and wear-free clearance seal technology, the linear motor has convinced the world of high reliability and efficiency by providing long life-time cryocooler system for space applications [3]. By coupling the TASHE with a linear alternator, it can be further developed as a thermoacoustic Stirling electric generator (TASEG), which potentially has advantages on high reliability and efficiency.

So far, three kinds of TASEG have been developed. The first one is the traditional TASEG, which uses the traditional thermoacoustic Stirling heat engine to drive linear alternator with or without a resonance tube. In 2004, Backhaus and Swift developed such a TASEG without a resonance tube [4]. In their experiments, an electric power of 58 W with 15% heat-to-electric conversion efficiency was achieved. After improvement, 70 W electric power was obtained with 16.8% thermal-to-electric efficiency [5]. To solve the coupling problem between the engine and alternator, our group proposed a TASEG with a resonance tube to improve the coupling relationship between the engine and the alternator. In 2008, a prototype of 100 W-class TASEG was developed [6]. In 2011, we obtained a maximum electric power of 481 W and a maximum thermal-to-electric efficiency of 15% in the experiments by designing a new TASEG [7] and a solar-powered system was also demonstrated [8]. After further investigation on the coupling rule, a maximum electric power of 1043 W with 18.6% heat-to-electric efficiency was achieved in 2012 [9]. In 2013, Sun developed such a TASEG with a maximum electric power of 345.3 W with about 10% efficiency [10]. In those work, both the long resonance tube (about 5 m long for 70–80 Hz working frequency) and the long feedback tube (about 1 m long) are required in the system. Thereby, the power density is very low and not suitable for

application in the future. The second one is acoustically resonant type TASEG, which has a loop configuration with one wavelength. The engine was proposed in 1979 by Ceperley [11] and verified by Yazaki in 1998 [12]. By increasing the acoustic impedance at the regenerator, Yu et al. developed an acoustically resonant TASEG with two regenerators in 2010 and 2012 [13,14]. The Aster-thermoacoustics company also developed such a TASEG [15]. However, the purpose of their systems is to utilize the low-grade thermal energy and the power density is not very high.

In order to improve the system power density and capacity, this paper proposed the third type of TASEG named double-acting thermoacoustic Stirling electric generator (DATASEG). In the DATASEG, at least three TASHEs and three linear alternators are connected end-to-end to build a three-cylinder double-acting system. Each linear alternator has two pistons, one act as an expansion piston to receive the acoustic work output from one engine, the other act as a compression piston to provide acoustic work to the regenerator of next engine. The work difference between the expansion piston and compression piston can be converted to electricity by the alternator. Thus, the resonance tube and feedback tube are no longer necessary, because their functions can be realized by the alternators. And the working frequency is determined by the system itself. Naturally, the moving phase angle between any two adjacent alternators equals to 360° divided by the number of the engines or alternators, for instance, 120° for three-cylinder system and 90° for four-cylinder system. Theoretically, the three or four-cylinder system can achieve a better performance and is most used in designing such a system. In order to verify this idea, a 3 kW DATASEG prototype is designed, built and tested in this paper.

In the following sections, the schematic and photograph of the system presented first as well as the typical parameters. Secondly,

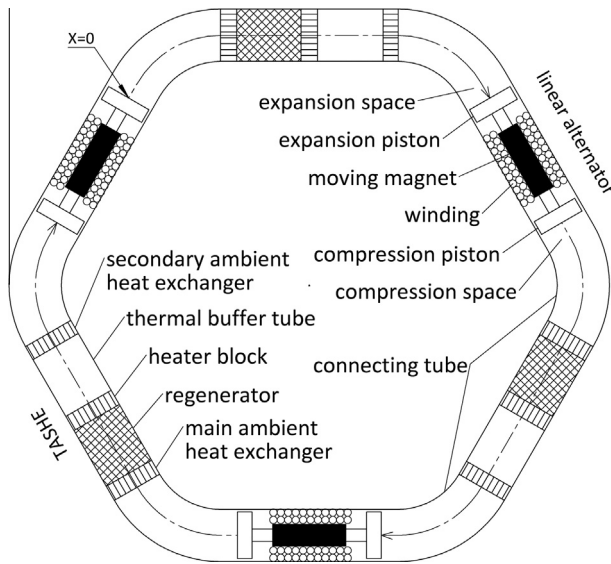


Fig. 1. Schematic of DATASEG.

numerical modelling is introduced and used to design the system. After that, testing results are presented and analyzed. Lastly, some conclusions are made.

2. System configuration

Fig. 1 demonstrates the schematic of the prototype of a three-cylinder double-acting thermoacoustic Stirling electric generator, which consists of three TASHes and linear alternators. Each TASH consists of a main ambient heat exchanger, a regenerator, a heater block, a thermal buffer tube and a secondary ambient heat exchanger. The heater block is used to input heating power to the system. The ambient heat exchangers are cooled by water. The hot-end heat exchanger and main ambient heat exchanger provide a temperature gradient along the regenerator, which is required by the thermoacoustic conversion effect. The thermal buffer tube and the secondary ambient heat exchanger are used to prevent the expansion piston of the alternator from operating within the hot environment and improve the reliability of the alternator. The linear alternator is a moving magnet type motor, which consist of a stator winding, a moving magnet connected with an expansion piston and a compression piston. The moving parts are usually suspended by flexure bearings or gas bearings to realize wear-free and oil-free clearance seal. During the experiments, after the heating power is inputted into the heater block, the system starts oscillation when the axial temperature gradient of the regenerator exceeds the critical value, and the pressure wave drives the pistons and magnets of the linear alternators to generate the electricity in the windings.

Here, in order to verify the design methodology, a prototype of 3 kW DATASEG is designed, fabricated and tested. In this prototype, the engines and alternators are connected in series and a loop configuration is constructed for structure simplicity. Before the engine design, 1 kW class linear alternators have been designed and fabricated first. The electrical and mechanical parameters of the alternators are described in Table 1. The linear alternators have a displacement limitation of 6.5 mm and a winding current limitation of about 5.5 A. Based on the numerical approach introduced in the next section, the engines can be designed and the optimized parameters are presented in Table 2.

According to these parameters, the 3 kW DATASEG was built and the photograph is presented in Fig. 2. The setup includes three

Table 1

Electrical and mechanical parameters of the alternator.

Piston diameter (mm)	75
Transduction coefficient (N A^{-1})	115
Winding resistance (Ω)	2.8
Winding inductance (mH)	218
Moving mass (kg)	2.45
Spring stiffness (kN m^{-1})	125
Mechanical damping coefficient (N s m^{-1})	50

Table 2

Structure parameters of the engine.

Component	Parameter
Compression space	50 cc
Connecting tube	$d = 50 \text{ mm}$, $l = 90 \text{ mm}$
Main ambient heat exchanger	$d = 50 \text{ mm}$, $l = 40 \text{ mm}$
Regenerator	$d = 50 \text{ mm}$, $l = 75 \text{ mm}$, 150 mesh stainless screen
Heater block	$d = 50 \text{ mm}$, $l = 50 \text{ mm}$
Thermal buffer tube	$d = 50 \text{ mm}$, $l = 75 \text{ mm}$
Secondary ambient heat exchanger	$d = 50 \text{ mm}$, $l = 40 \text{ mm}$
Connecting tube	$d = 50 \text{ mm}$, $l = 90 \text{ mm}$
Expansion space	50 cc

thermoacoustic heat engines and three linear alternators. In each heater block, twenty heat cartridges are used to supply a maximum heating power of about 4 kW. The hot part of the engine, including the regenerator, the heater block and the thermal buffer tube are covered by thermal insulation materials in the experiments. The two ambient heat exchangers are cooled by 30°C water. Three rheostats are used as load resistances of the alternators to consume the electric power generated by the alternator. To avoid damage of the linear alternator caused by a sudden startup at a high heating temperature, one alternator keeps running as a compressor to provide continual excitation until the system starts up. During the experiments, the heating power of three heater blocks are measured by a three-phase wattmeter. The heating temperatures are measured by K type thermocouples. The voltage and current waves of the rheostats are measured by voltage and current sensors (model P5200 and TCP312) from Tektronix. High-precision dynamic pressure sensors (model 113B26) from PCB Piezotronics are installed at the compression and expansion spaces. Three accelerometers (model M353B14) from PCB Piezo-

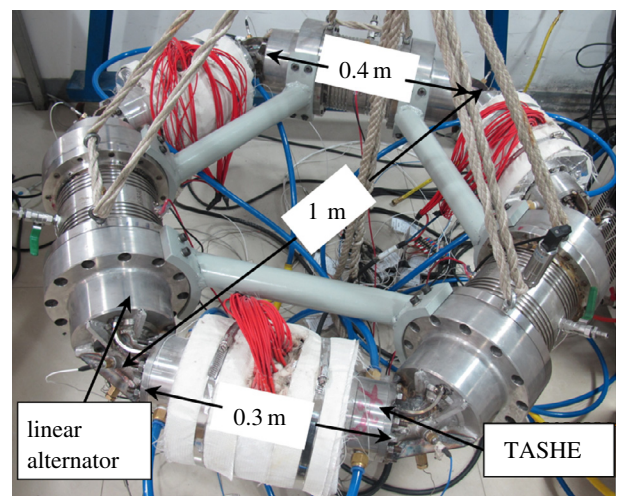


Fig. 2. Photograph of 3 kW three-cylinder DATAEG prototype.

tronics are used to monitor the movements of the pistons. All signals are collected by PXI-4472B from National Instruments, which can perform simultaneous data acquisition with high sample rate.

3. Thermodynamic design

In order to obtain the thermodynamic parameters of the engine, numerical simulation should be performed first. Here, we use one dimensional cross-sectional area averaged model, which is proved to be an efficient approach with promising accuracy in designing a thermoacoustic system. The computational model is based on the classic thermoacoustic theory founded by Rott [16] and advanced by Xiao [17–19] and Swift [20]. According to the theory, a program has been developed by our group [21,22].

3.1. Modelling

According to Rott's acoustic approximation of the momentum and continuity equations, control equations for arbitrary shape and size of flow channels can be rewritten as

$$\frac{dp}{dx} = -\frac{i\omega\rho}{A(1-f_v)}U \quad (1)$$

$$\frac{dU}{dx} = -\frac{i\omega A}{\gamma P_0}(1+(\gamma-1)f_\kappa)p + \frac{(f_\kappa-f_v)}{(1-f_v)(1-\sigma)}\frac{1}{T}\frac{dT}{dx}U \quad (2)$$

The total energy equation is

$$H = \frac{1}{2}\text{Re}\left[pU^*\left(1 - \frac{f_\kappa-f_v^*}{(1+\sigma)(1-f_v^*)}\right)\right] + \frac{\rho c_p|U|^2}{2A\omega(1-\sigma^2)|1-f_v|^2}\text{Im}[f_\kappa + \sigma f_v^*]\frac{dT}{dx} - (Ak + A_s k_s)\frac{dT}{dx} \quad (3)$$

Based on the energy balance, we have

$$\frac{dH}{dx} = Q \quad (4)$$

In the calculation, the physical model of the engine will be divided into a certain number of cells, see Fig. 3. Solving Eqs. (1) and (2), we can obtain the relationship of two neighboring cells in the form of a transfer matrix. For conciseness, we have:

$$\begin{bmatrix} p_{(n+1)} \\ U_{(n+1)} \end{bmatrix} = \begin{bmatrix} \text{TM}_{11} & \text{TM}_{12} \\ \text{TM}_{21} & \text{TM}_{22} \end{bmatrix}_{(n)} \cdot \begin{bmatrix} p_{(n)} \\ U_{(n)} \end{bmatrix}$$

where n is the cell number. TM is a 2×2 transfer matrix with complex elements, which are functions of the fluid passage geometry, frequency, average pressure, volume flow rate, temperature distribution, etc. Eqs. (3) and (4) can be written with a first-order accuracy for each cell in a discrete form:

$$B_{(n,n-1)}T_{(n-1)} + B_{(n,n)} \cdot T_{(n)} + B_{(n,n+1)} \cdot T_{(n+1)} = BB_{(n)} \quad (6)$$

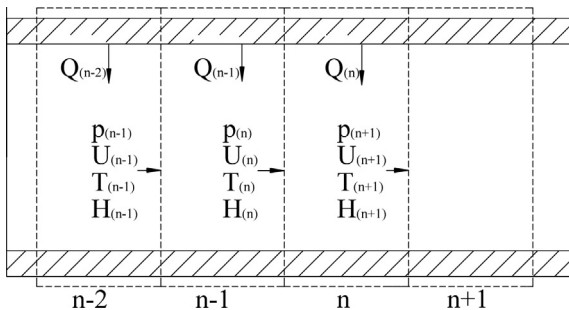


Fig. 3. Schematic of numerical model.

where B is a $N \times N$ matrix with complex coefficients related to the fluid passage geometry, frequency, average pressure, volume flow rate, temperature distribution, etc. BB is a $N \times 1$ matrix. Obviously, the temperature distribution can be easily solved using linear algebra algorithms. According to Eqs. (5) and (6), the volume velocity and the temperature are dependent on each other, iteration process is necessary to obtain new distributions of p , U and T . Before the calculation, the working frequency and the temperature should be guessed as initial condition. And the control equations of the alternator listed below are used as boundary conditions.

$$\frac{\tau}{A_{pe}}U_e - (R + r + i\omega L)I = 0 \quad (7)$$

$$\frac{(R_m + i(\omega M - K/\omega))}{A_{pe}^2}U_e + \frac{\tau}{A_{pe}}I = p_e - p_c \frac{A_{pc}}{A_{pe}} \quad (8)$$

where p_e and p_c represent the pressure amplitude in the expansion and compression space, respectively. A_{pe} and A_{pc} are cross-sectional area of expansion and compression piston. R is the load resistance connected with the alternator to consume the generated electric power.

The electric power and the acoustic power are calculated respectively by

$$W_e = \frac{1}{2}|V||I|\cos(\theta_{VI}) \quad (9)$$

$$W_a = \frac{1}{2}|p||U|\cos(\theta_{pU}) \quad (10)$$

The thermal-to-electric conversion efficiency of the system and the acoustic-to-electric conversion efficiency of the alternator are defined as

$$\eta_{te} = \frac{W_e}{Q_h} \quad (11)$$

$$\eta_{ae} = \frac{W_e}{W_a} \quad (12)$$

The piston displacement and the volume velocity have the relationship with the acceleration as

$$x = \frac{a}{\omega^2} \quad (13)$$

$$U = \frac{a}{\omega}A_p \quad (14)$$

3.2. Numerical results

In order to further understand the thermodynamic characteristics of the DATASEG, axial distributions of key parameters are presented in Figs. 4–7. In these figures, the X axis begins at the compression piston of one alternator and goes clockwise as demonstrated in Fig. 1. Because lumped parameters of the alternators are used in the simulation, the X axis does not include the alternator length. The working frequency is 84.5 Hz determined by the system itself. In the simulation, 5.0 MPa mean pressure helium is used as the working gas. Because the system performances under a high temperature are most concerned, the heating temperature is fixed at 650 °C. The load resistance is fixed at 68.5 Ω and the piston displacement is fixed at 6.5 mm.

Fig. 4 demonstrates the axial distribution of gas temperature. From the temperature curve, the gas temperature is increased in the regenerator and decreased in the thermal buffer tube. The temperature difference caused by heat transfer also can be found in the heater block and ambient heat exchangers. From the distributions

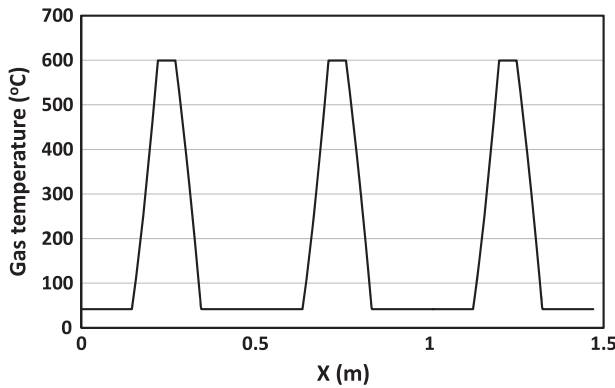


Fig. 4. Axial distribution of gas temperature.

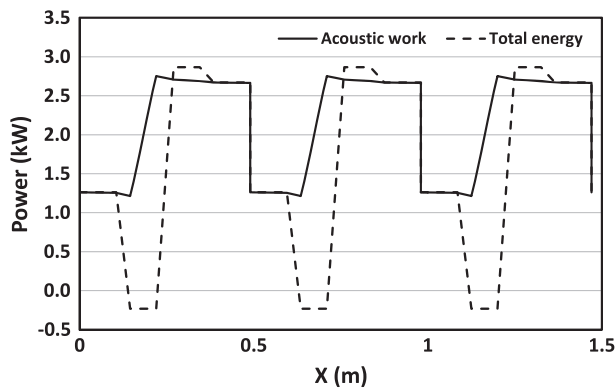


Fig. 5. Axial distributions of acoustic work and total energy.

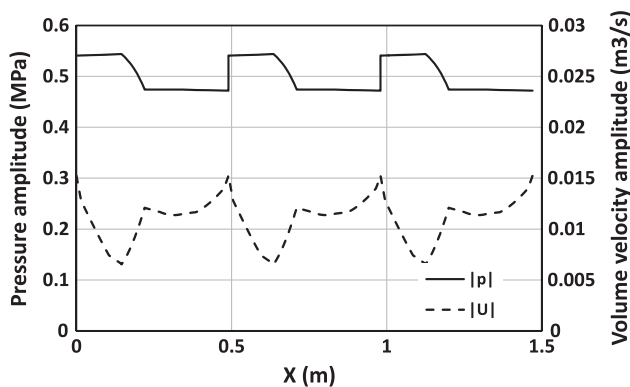


Fig. 6. Axial distributions of pressure amplitude and volume velocity amplitude.

of acoustic work and total energy in Fig. 5, about 1.26 kW acoustic work is inputted to the engine by the compression piston of one alternator. In the main ambient heat exchanger, the total energy decreases from 1.26 kW to -0.23 kW. Since the total energy is defined as the sum of the acoustic work and the heat exchange, it means that about 1.49 kW heat is released to the main ambient heat exchanger by the working gas. And the acoustic work is dissipated about 47 W in the main ambient exchanger. In the regenerator, the acoustic work is then amplified to 2.75 kW while the total energy remains unchanged, which demonstrates the energy conversion between heat and acoustic work. In the heater block, the total energy is increased from -0.23 kW to 2.87 kW and 3.1 kW heating power is absorbed by the working gas. In the secondary ambient heat exchanger, about 0.2 kW heat is released and the

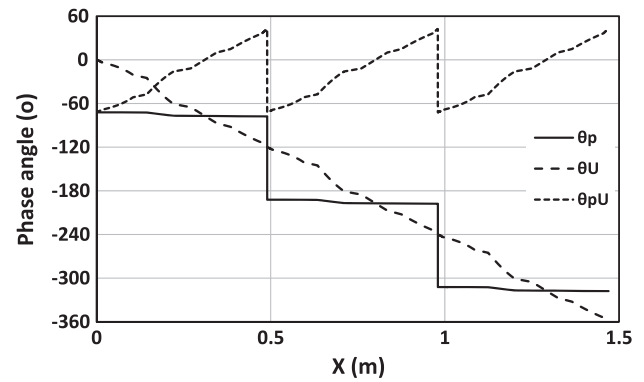


Fig. 7. Axial distributions of phase angle of pressure wave and volume velocity and the difference between them.

total energy is slightly decreased. At the outlet of the engine, the acoustic work is separated into two parts by another alternator, 1.4 kW acoustic work is converted into electricity and the remaining 1.26 kW acoustic work is transferred to the next engine by the compression piston. The computational output electric power for each alternators is 1.06 kW (3.18 kW for all alternators) and the thermal-to-electric conversion efficiency is 34.3%.

Figs. 6 and 7 show the distributions of the pressure and volume velocity amplitude as well as the phase angles. According to the phase angles distribution of the volume velocity, it can be seen that the volume velocity phase difference between the alternators at the two ends of each engine is 120° . Moreover, the pressure phase angle changes slightly along one engine. The phase angle difference between the pressure wave and volume velocity before the compression space and after expansion space are -72.2° and 42.4° , respectively.

The load resistance of the alternator plays an important role on the working state of the system, such as output electric power, efficiency, displacement and so on. At the beginning of the heating process, the resistance value of the load should be large to decrease the startup temperature. As the heating temperature is increased to 650°C , the load resistance should be decreased to a proper value to achieve the design performance. Figs. 8–10 demonstrate the influence of the load resistance on the system performance. In Fig. 8, the thermal-to-electric efficiency has an optimum value as the load resistance decreases from $150\ \Omega$ to $55\ \Omega$. However, according to the displacement curve in Fig. 9, the piston displacement will greatly exceed the allowed limit for a high load resistance if the heating temperature is fixed at 650°C . It also means that the heating temperature should be kept below 650°C until the load resistance is lower than about $70\ \Omega$. Due to a high

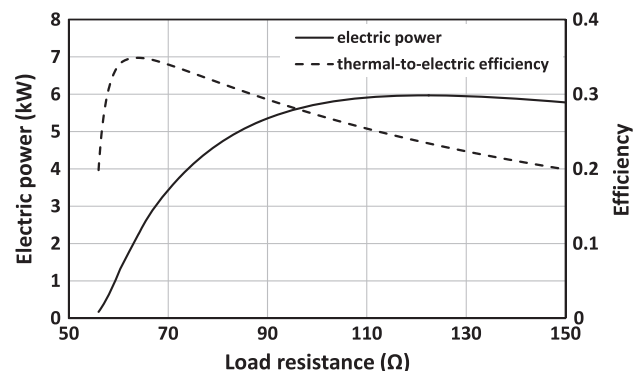


Fig. 8. The computational electric power and thermal-to-electric efficiency as functions of load resistance.

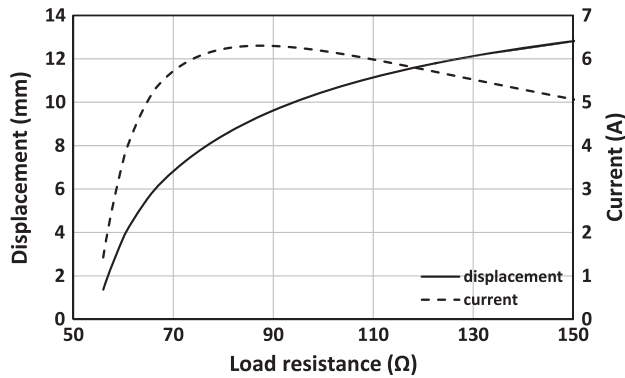


Fig. 9. The computational displacement and current of the alternator as functions of load resistance.

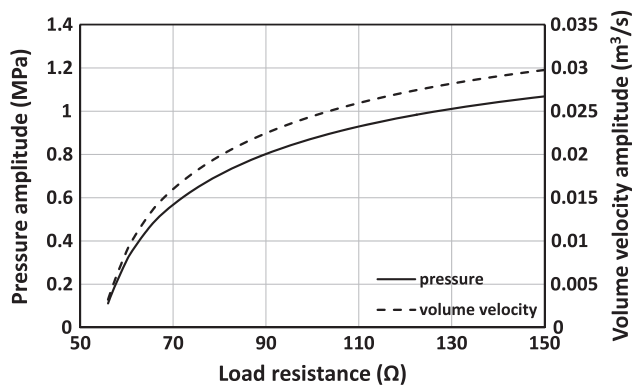


Fig. 10. The computational pressure amplitude and volume velocity amplitude as functions of load resistance.

mechanical loss caused by large piston displacement, the system efficiency is greatly decreased when the load resistance is increased to a high value. Meanwhile, the electric power can reach an optimum value where the efficiency is relatively low. So, one should balance between the efficiency and power density by choosing proper values of the pressure amplitude and volume velocity amplitude in the system design.

4. Experimental results and discussion

After the system setup was built, experiments were performed to test the system performance. Figs. 11–13 present the experimental data of piston phase angle, electric power, efficiency, heating temperature and working frequency as functions of load resistance. In these experiments, 5.0 MPa pressurized helium was used as working gas. The heating power was fixed at about 3 kW for each heater block and the heating temperatures of the heater blocks were controlled below 650 °C. Fig. 11 shows the moving phase angles of three pistons. It can be seen that the phase angle difference between two adjacent pistons is about 120°, which agrees with the simulation results in Fig. 7. As demonstrated in Fig. 12, it presents the electric powers of three alternators and the thermal-to-electric efficiency of the system. In the experiments, a maximum total electric power of 1.57 kW with a thermal-to-electric efficiency of 16.8% was achieved. And the working frequency is about 86 Hz in Fig. 13, which is very close to the design value.

As shown in Figs. 11–13, two important things can be found. Firstly, the experimental data is far below our expectation compared with the computational data in Fig. 8. One reason is that

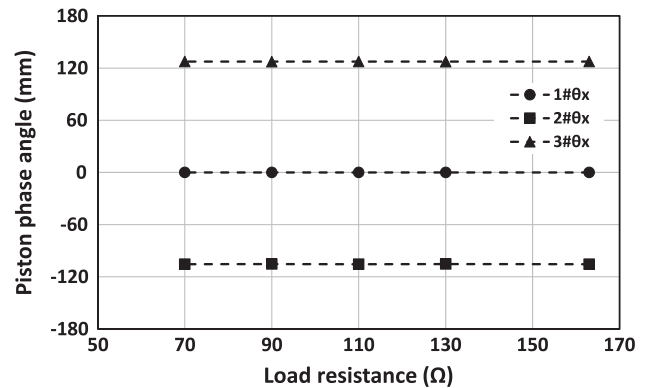


Fig. 11. Experimental results of piston phase angle.

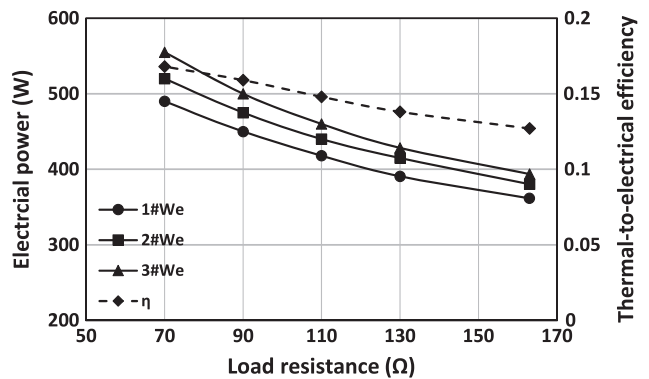


Fig. 12. Experimental results of electric power and thermo-to-electric efficiency.

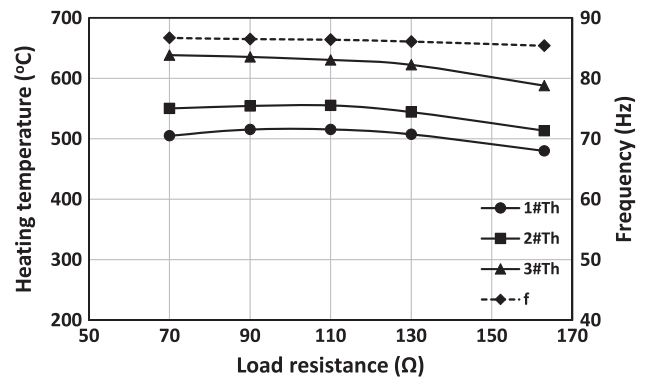


Fig. 13. Experimental results of heating temperature and working frequency.

the maximum piston displacement in the experiments is decreased to 6 mm to avoid overstroke and the output power is also decreased. The permitted displacement of the piston is 7 mm, however we found that the equilibrium position of the piston shifted about 1 mm with larger piston stroke during operation and the actual allowable displacement must be reduced to 6 mm. The possible reason may be related with the asymmetric clearance seal leakage of the piston [20]. Another reason is the high mechanical damping coefficient of the alternator. The mechanical damping loss consists of the friction loss between the piston and the cylinder and the clearance seal loss. We have developed a testing setup to measure the mechanical damping coefficient separately without the engines and the value is presented in Table 1. However, during the experiments, we found that gas leakage easily happened at the welding place of two ends of the connecting tubes even when the

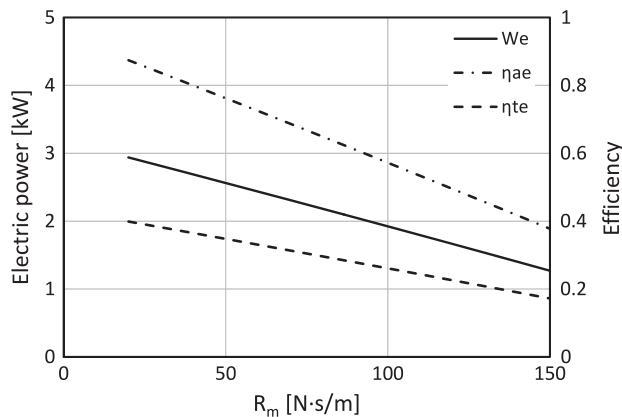


Fig. 14. Influence of the mechanical damping coefficient on the system performance.

wall thickness is much higher than required value. We believe that the large thermal stress was responsible for this phenomenon. Thereby, the cylinder wall must be deformed under this stress. The clearance between the piston and cylinder wall is about 10–15 μm . Thus, friction between the piston and cylinder wall easily occurs. This can be proved by the severe abrasion marks on the cylinder wall and piston side wall. According to our experience, the mechanical damping coefficient can be dramatically increased by friction. Fig. 14 shows the influence of the mechanical damping coefficient on the system performance. From Fig. 14, the electric power and the acoustic-to-electric conversion efficiency of the alternator are greatly decreased as well as the thermal-to-electric conversion efficiency of whole system because of the increase of the mechanical damping coefficient. However, the real mechanical damping coefficient is difficult to be measured during operation.

Secondly, from these figures, we note that the heating temperature differences between the heater blocks and the electric power differences between the alternators are very large, which demonstrate a serious system performance difference. This difference also caused the performance degradation of the system, for example, engine performance with 500 $^{\circ}\text{C}$ heating temperature is much lower than it with 650 $^{\circ}\text{C}$ heating temperature. However, the reasons for this phenomena are complicate. We are now developing an experimental setup to evaluate the performance difference of the engines with the same working conditions [23]. Moreover, Structure improvement is under way to avoid high and nonuniform mechanical damping coefficient. More experimental data will be reported in the future.

5. Conclusions

This work introduces a novel double-acting thermoacoustic Stirling electric generator capable of converting external heat to electricity. Compared with the traditional TASEG, it potentially has advantages on power density and efficiency by employing at least three engines and alternators and eliminating the resonance and feedback tubes. To verify the idea, a prototype of 3 kW three-cylinder double-acting thermoacoustic Stirling electric generator was designed, built and tested in this paper. Firstly, numerical simulation is performed to obtain optimum structure parameters of the engine. Additionally, distributions of temperature, phase angle, amplitude of pressure and volume velocity, acoustic power and total energy are presented to understand the energy conversion characteristic of the system. Then, a simple loop configuration experimental setup was built. In the preliminary

experiments, a maximum electric power of 1.57 kW with 16.8% thermal-to-electric conversion efficiency was obtained. Due to the reduction of permitted displacement caused by equilibrium position shifting of the piston and the high mechanical damping coefficient caused by cylinder deformation under high thermal stress, the system performance was greatly decreased. Moreover, the performance differences of the system are significant, which is also responsible for the system performance degradation. In the future, research work will focus on identification of the engines and alternators performance differences as well as structure improvement to avoid the cylinder deformation.

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